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CALCULATIONS FOR AXIAL COMPRESSOR BLADING WITH UNIFORM INLET EN--ETC(U)
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Calculations for Axial Compressor Blading with Uniform Inlet Enthalpy and Radial Enthalpy Gradient

W. Schlachter

May 1981

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NAVAL POSTGRADUATE SCHOOL

Monterey, California

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This previously unreported work was carried out while the author was Naval Air Systems Command Visiting Research Professor in Aeronautics at the Naval Postgraduate School during 1971-72. The significance of the results is unchanged and the present report is issued to serve as a proper reference for future work. The results were presented at a NAVAIR Contractor's Meeting held in Washington, DC, in December 1971.

This report was prepared from the original draft of W. Schlachter (currently Dept T-T, Brown Boveri & Co. Ltd., Baden, Switzerland) with Dr. Schlachter's concurrence

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

A computer program was used to calculate the radial distribution of flow parameters in an axial compressor stage designed to have a symmetrical velocity diagram at the mean radius and particular variations of reaction from hub to tip. Uniform energy addition was assumed to occur in the rotor. Both cases of uniform enthalpy and uniform radial enthalpy gradient at the entrance to the stage were considered. Advantages were found in the selection of

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20. ABSTRACT (continued)

fully symmetric blading and in the use of the inlet enthalpy gradient as a design parameter.

CALCULATIONS FOR AXIAL COMPRESSOR BLADING
WITH UNIFORM INLET ENTHALPY AND RADIAL
ENTHALPY GRADIENT

by

W. Schlachter

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1. Introduction

In 1970, new blading was designed for the large 3 stage axial research compressor (36 inches O.D by 21 6 inches I.D) at the Turbopropulsion Laboratory at the Naval Postgraduate School. (Ref 1). The design was reported in Ref 2 and the installation of the compressor in Ref 3. A program to examine the effects of tip clearance gap on the performance of different designs of compressor blading was proposed and the new blading was intended to provide a base data-set for blading of a then conventional type. The new blading was designed for uniform energy input along the radius and employed high aerodynamic loading. The design was however, of the symmetrical type, for reasons discussed in detail in Ref 2 and Ref 4. Other blading sets also available for, and previously used in the compressor were free vortex, forced vortex (or solid body) and constant angle designs, but were much more lightly loaded.

The intended research program was interrupted for a period of more than seven years. Recently, work was resumed and the new compressor blading was completed and one stage was installed. The present report documents the results of calculations carried out but not reported in the earlier phase of the program completed in 1971.

Calculations were made to determine the radial equilibrium conditions for different bladings designed to have uniform energy input along the radius, with and without a specific radial energy gradient as an inlet condition to the stage. The calculations were made for a single stage (in a repeating stage design) using the computer program given in Ref 5.

2. Approach

Three-dimensional flow calculations in turbomachines are generally carried out in a simplified way. The flow is assumed to be axisymmetric and steady, and the equation of motion is solved between the blade rows only. The influences of the streamline curvature, of the entropy and enthalpy gradients along the blade height are sometimes but not always taken into account.

The direct design method usually applied for axial compressor stages assumes that the blades designed with 2-dimensional cascade data produce the calculated flow angles. This method has been generally successful for moderate blade loadings. It was applied in the design of blading reported in Ref 2 which has a high aerodynamic loading. The method for the flow calculations used in Ref 2 is given in Ref 4. A computer program, based on the same method, but which takes into account the influences mentioned above, as well as the distribution of the losses along the blade height, was reported in Ref 5. Using the program (and specified loss models) in Ref 5, calculations were carried out using the geometrical data of the blading. In the first series, the hand-calculations of Ref 2 were checked. The agreement was found to be good. In the second series, 7 cases with different degrees of reaction over the blade height were calculated in order to survey other possible types of bladings. The goal was to find a blading having:

- uniform inlet velocities relative to the rotor as well as to the stator, which were small compared with the peripheral velocity

- diffusion factors for all profile sections as high as the necessary stall limit permitted.

In the third series, the effect of a further degree of freedom was investigated; namely, the influence of imposed enthalpy gradients. This influence has been found to be considerable, (Ref 4).

Results of the second and third series of calculations are presented here. They provide information, which, together with the experiments carried out with the symmetrical blading, help determine what type of blading should be selected for a second phase of the test program.

3. Procedure

The type of stage considered and the notation for the following discussion are given in Fig 1 and Fig 2. The degree of reaction is assumed to be 50% at the mean radius R_m. The degree of reaction is defined here as

$$r^* = 1 - \frac{V_{u1} + V_{u2}}{\omega(R_1 + R_2)} \tag{1}$$

where ω is the angular velocity.

The energy input

$$\Delta H = \omega(R_2 V_{u2} - R_1 V_{u1}) = \omega K \tag{2}$$

is assumed to be constant over the blade height in the present calculations. A variety of stages is analyzed with the general distribution of the peripheral components of the absolute velocity ahead of the rotor being given by

$$V_{u1} = A \frac{R_m}{R} + B + C \frac{R}{R_m} \tag{3}$$

Eqs. (1), (2) and (3) give the relationship between the coefficients A, B and C with the degree of reaction

$$r^*(R) = 1 - \frac{1}{\omega R_m} \left[\left(\frac{R_m}{R} \right)^2 \left(A + \frac{K}{2R_m} \right) + B \frac{R_m}{R} + C \right] \tag{4}$$

With the assumed conditions, each special case can be represented by a point in a rectangular coordinate system with A, B and C measured along the axes. As shown in Fig 3, all possible points lie in a plane. In addition to the three more familiar blade types, i.e.

free vortex; $A = \frac{1}{2} (\omega R_m - \frac{K}{R_m})$, $B = C = 0$

V_{u1} = const.; $B = \frac{1}{2} (\omega R_m - \frac{K}{R_m})$, $A = C = 0$

forced vortex; $C = \frac{1}{2} (\omega R_m - \frac{K}{R_m})$, $A = B = 0$

the three other cases identified in Fig 3 were investigated here. It was thought that these examples would illustrate the range of possibilities that exist with and without imposed enthalpy gradients.

The distribution of the degree of reaction over the blade height is shown in Fig 4. The same cases were investigated with an imposed enthalpy gradient which is assumed to be constant over the blade height. It can therefore be represented by

$$\frac{\partial H}{\partial R} = 2\zeta \frac{\omega K}{R_t - R_h} \tag{5}$$

where the factor ζ itself gives the magnitude of the gradient. The two assumptions made above (ΔH = constant over blade height, $r^* = 0.50$ at $R = R_m$) are retained because we are considering a so-called standard stage. The enthalpy gradient has to be produced by an entrance stage. Selected values for the present calculations were taken between $\zeta = -0.15$ (negative enthalpy gradient) and $\zeta = +0.15$ (positive gradient).

4. Results for Uniform Enthalpy ($\zeta = 0$)

Figs. 5a and 5b show the distribution of the axial velocity component upstream and downstream the rotor. At station 1, the velocity profile of the free vortex flow is nearly uniform as one would expect in a flow without enthalpy and entropy gradients. Since the losses produced by the rotor are not uniformly distributed over the blade height, there is an entropy gradient at station 2. This gradient is large near the tip due to end losses and tip clearance losses. The considerable influence of the entropy gradient is seen best in the velocity profile of the free vortex flow, Fig 5b which is not uniform any more.

The relative inlet velocity ahead of the rotor is plotted in Fig. 6. It shows the well known result that bladings as free vortex type, V_{u1} = constant, case $\boxed{2}$, that means bladings with a low coefficient C in Eq. (4), are not suited for axial compressors of high flow rate and high pressure rise per stage. For a given maximum Mach number, these stages must be operated at lower peripheral velocities due to the conditions at the rotor tip. However, bladings with a high coefficient C (symmetrical type, cases $\boxed{1}$ and $\boxed{3}$, Fig. 3) are much more favorable in this respect.

Figs. 7a and 7b show the increase of the static pressure in the rotor and in the stator, respectively. The pressure rise is shown in dimensionless form using the quantity $\rho(\omega R_m)^2$. The bladings with high coefficients C have a lower static pressure difference at the rotor tip and at the stator hub, respectively, than the other cases. This would be expected to reduce the importance of tip clearance effects.

The NASA diffusion factors are shown in Figs. 8a and 8b. A blading with a lower degree of reaction near the rotor tip than that obtained in case $\boxed{3}$ would yield a large diffusion factor at the tip of the rotor as well as at the tip of the stator. Therefore, case $\boxed{3}$ represents the other extreme to the vortex flow.

5. Results for Uniform Radial Enthalpy Gradient ($\zeta \neq 0$)

For comparison with the results for $\zeta = 0$, the same quantities are plotted in Figs. 9 - 12 assuming a positive enthalpy gradient with $\zeta = +0.15$. The free vortex flow is not irrotational any more as the non-uniform velocity profiles ahead and after the rotor show, (Figs. 9a and 9b). A positive enthalpy gradient tends to "equalize" the velocity profiles of bladings with high coefficient C, due to the fact that the pressure distribution ahead of the stage is different from that one would obtain with $\zeta = 0$. An almost uniform relative inlet velocity is obtained with a symmetrical blading, (Fig. 10.) It is clear that the pressure rise in the rotor at constant energy input and given distribution of the degree of reaction over the blade height is influenced very little by an enthalpy gradient. The difference is due to changes of the losses in the rotor, (Figs. 11 and 7a.)

A positive enthalpy gradient reduces the *blade loading* at the rotor tip as well as at the stator tip, (Figs. 11a and 11b.) The situation becomes worse at the stator hub where we have the stator blade gap. However, the most interesting cases (high coefficient C) are not critical in this respect, since the diffusion factors near the stator hub are at a low level, (Figs. 11b, 15a.) Figs. 13 - 15 show the influence of the imposed enthalpy gradient on the most interesting properties of the different blade types.

6. Concluding Remarks

According to these calculations, the selection of an imposed enthalpy gradient gives an additional degree of freedom in the design of axial compressor stages. Favorable flow patterns for particular applications can be obtained. In regard to the proposed research program, the symmetrical blading as selected in Ref. 2 for the first phase of experiments, turns out to be a good choice. This type of blading is found to be less Mach number limited by conditions at the outer radius. In addition, the increase of static pressure at the rotor tip is smaller than for the other bladings, except for case $\boxed{3}$. The blade loading at the critical stations at the rotor tip and at the stator hub, where the clearance losses occur, is not too excessive. The situation can be improved by an imposed positive enthalpy gradient of about $\zeta = 0.15$. This yields a favorable distribution of the diffusion factors, in the rotor as well as in the stator, Figs. 11a and 11b.) In addition, this blading has an almost constant relative inlet Mach number ahead of the rotor, (Fig. 10.)

Based on these calculations, the preliminary conclusion is that an entrance stage should be added which produces an enthalpy gradient of about $\zeta = 0.15$. The absolute peripheral velocity component downstream of this entrance stage has to be equal to that required ahead of a standard stage with symmetrical blading and constant energy input over the blade height. Such a blading has moderate changes of the flow angles over the blade height, (Fig. 16), and the changes of the absolute and relative flow angles, (Fig. 2,) are quite similar. Therefore, the blades have a reasonably small twist; hence they are favorable with respect to mechanical stresses.

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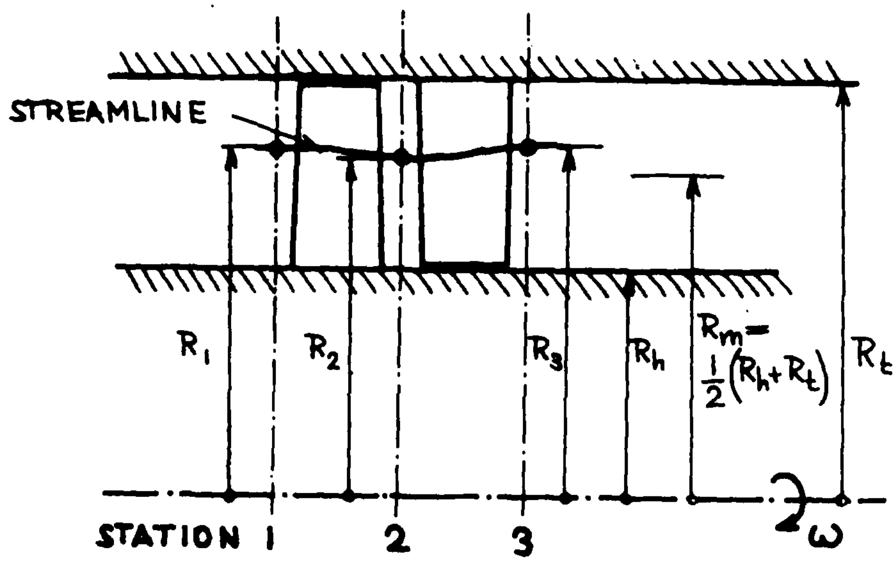

FIG. 1 AXIAL COMPRESSOR STAGE, CYLINDRICAL WALLS

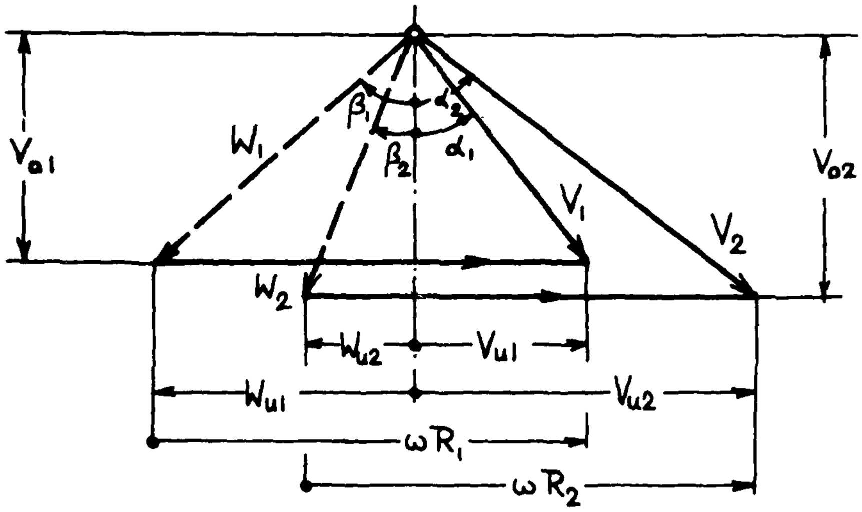

FIG. 2 VELOCITY DIAGRAM OF STAGE WITH FLOW ON ARBITRARY AXISYMMETRIC STREAM SURFACE

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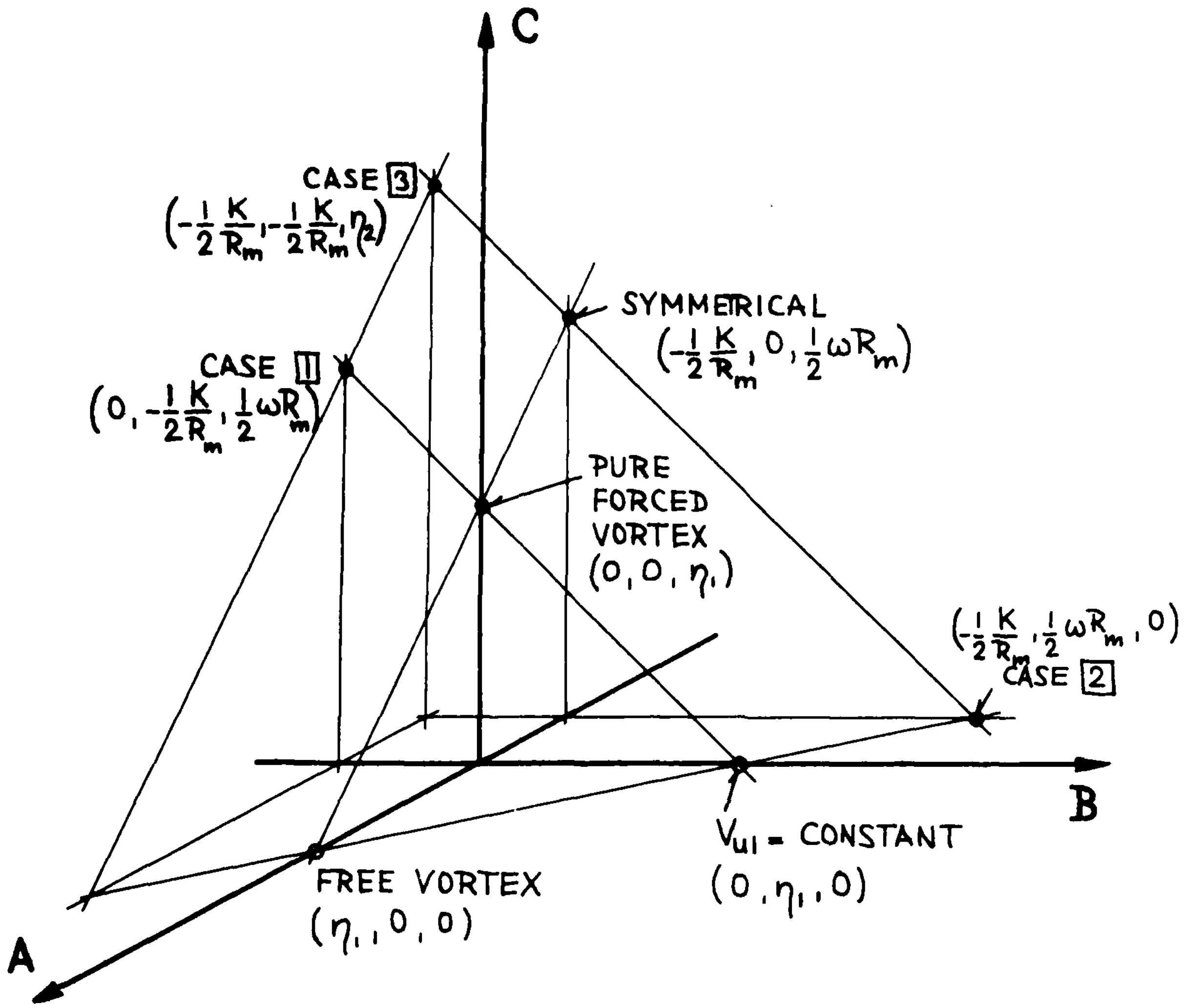

FIG. 3 REPRESENTATION OF INVESTIGATED CASES IN THE ABC - COORDINATE SYSTEM ($r^* = 0.5$ at $R = R_m$, CONSTANT ENERGY INPUT OVER ROTOR BLADE HEIGHT)

$$\eta_1 = \frac{1}{2}\left(\omega R_m - \frac{K}{R_m}\right) \quad , \quad \eta_2 = \frac{1}{2}\left(\omega R_m + \frac{K}{R_m}\right)$$

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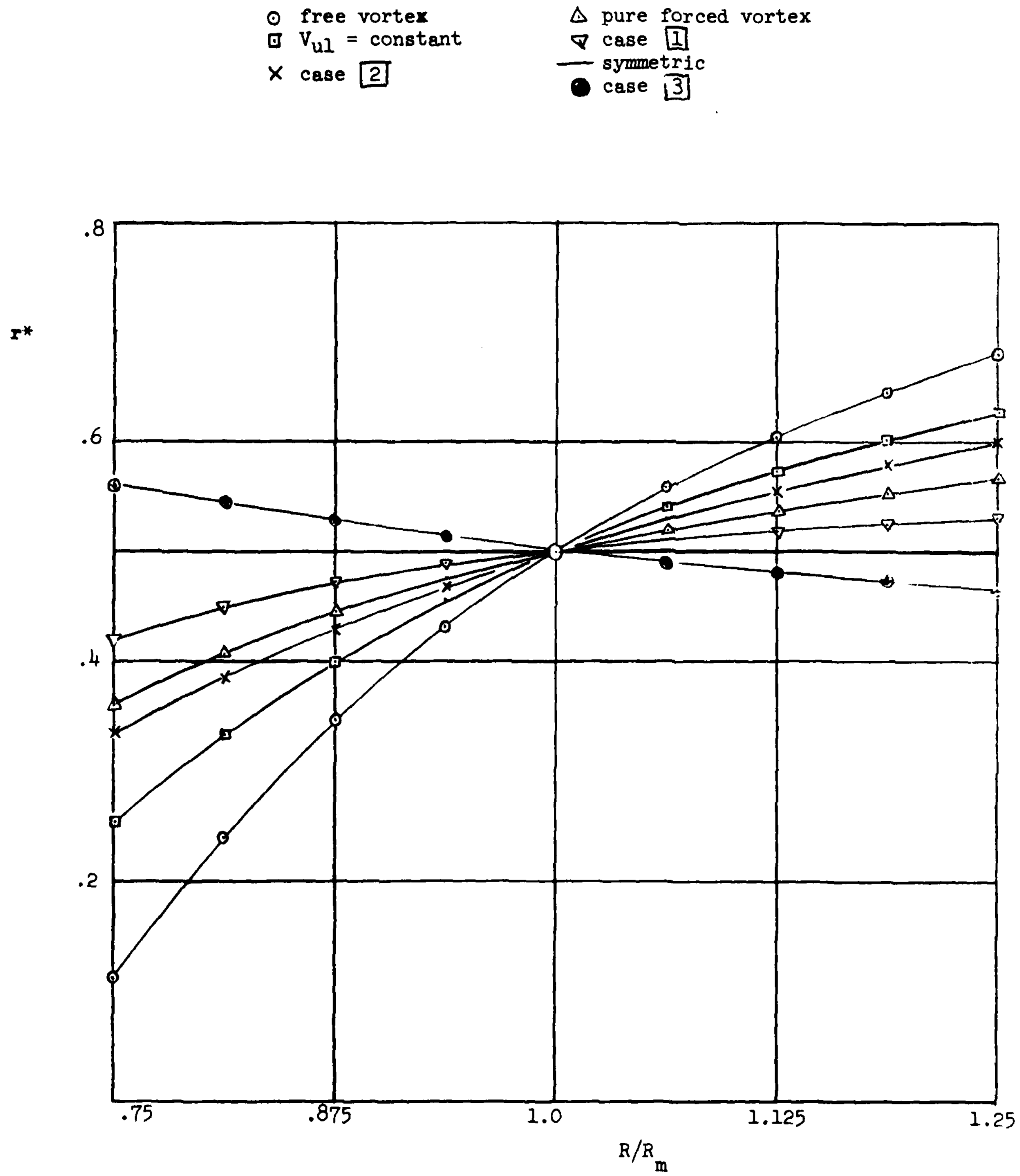
FIG. 4 KINEMATIC DEGREE OF REACTION OVER BLADE HEIGHT FOR DIFFERENT CASES



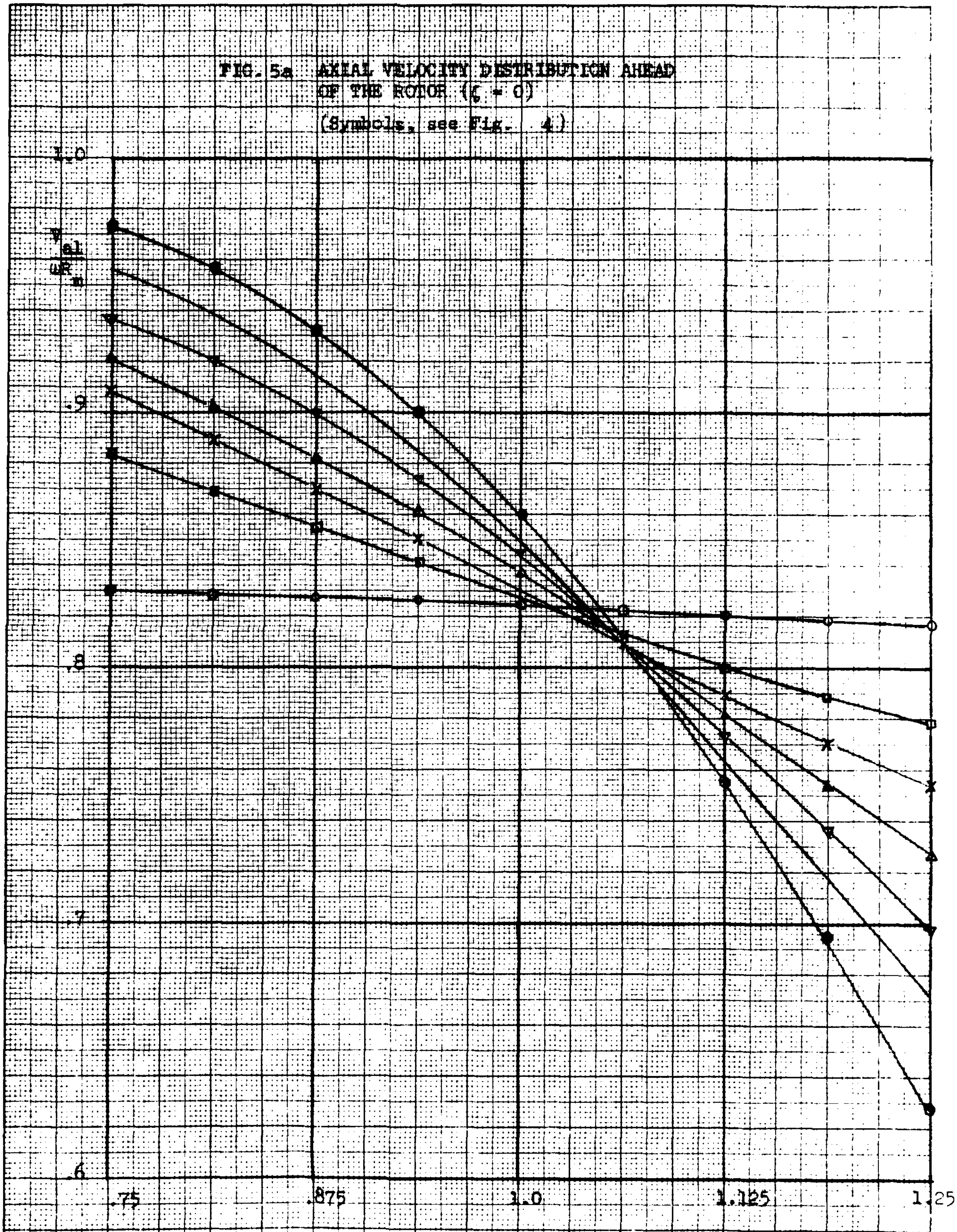


FIG. 5a AXIAL VELOCITY DISTRIBUTION AHEAD OF THE ROTOR (ζ = 0)
(Symbols, see Fig. 4)
1.0
.9
.8
.7
.6
.75
.875
1.0
1.125
1.25
R/R_m

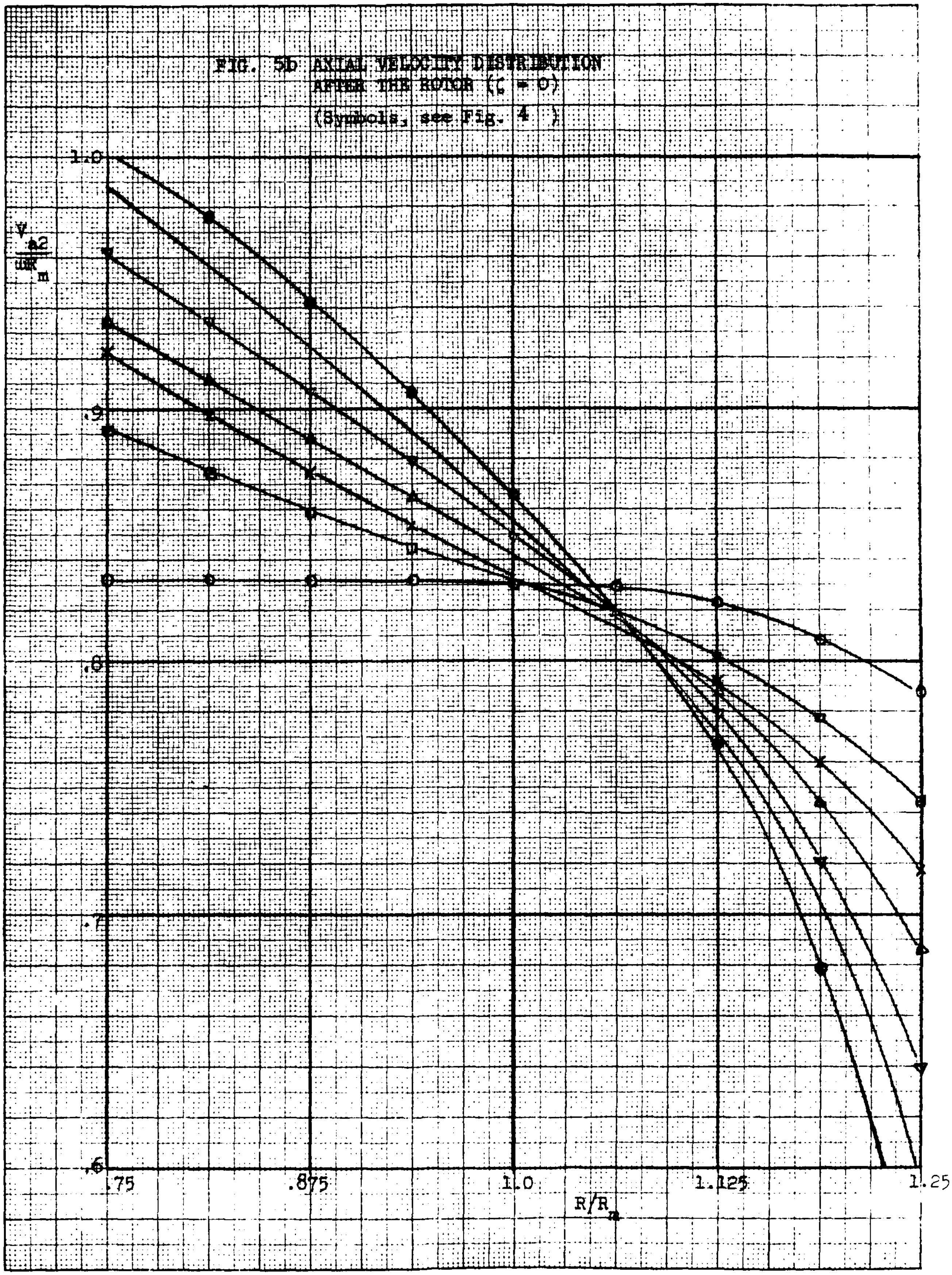


FIG. 5b AXIAL VELOCITY DISTRIBUTION AFTER THE ROTOR (ζ = 0)
(Symbols, see Fig. 4)

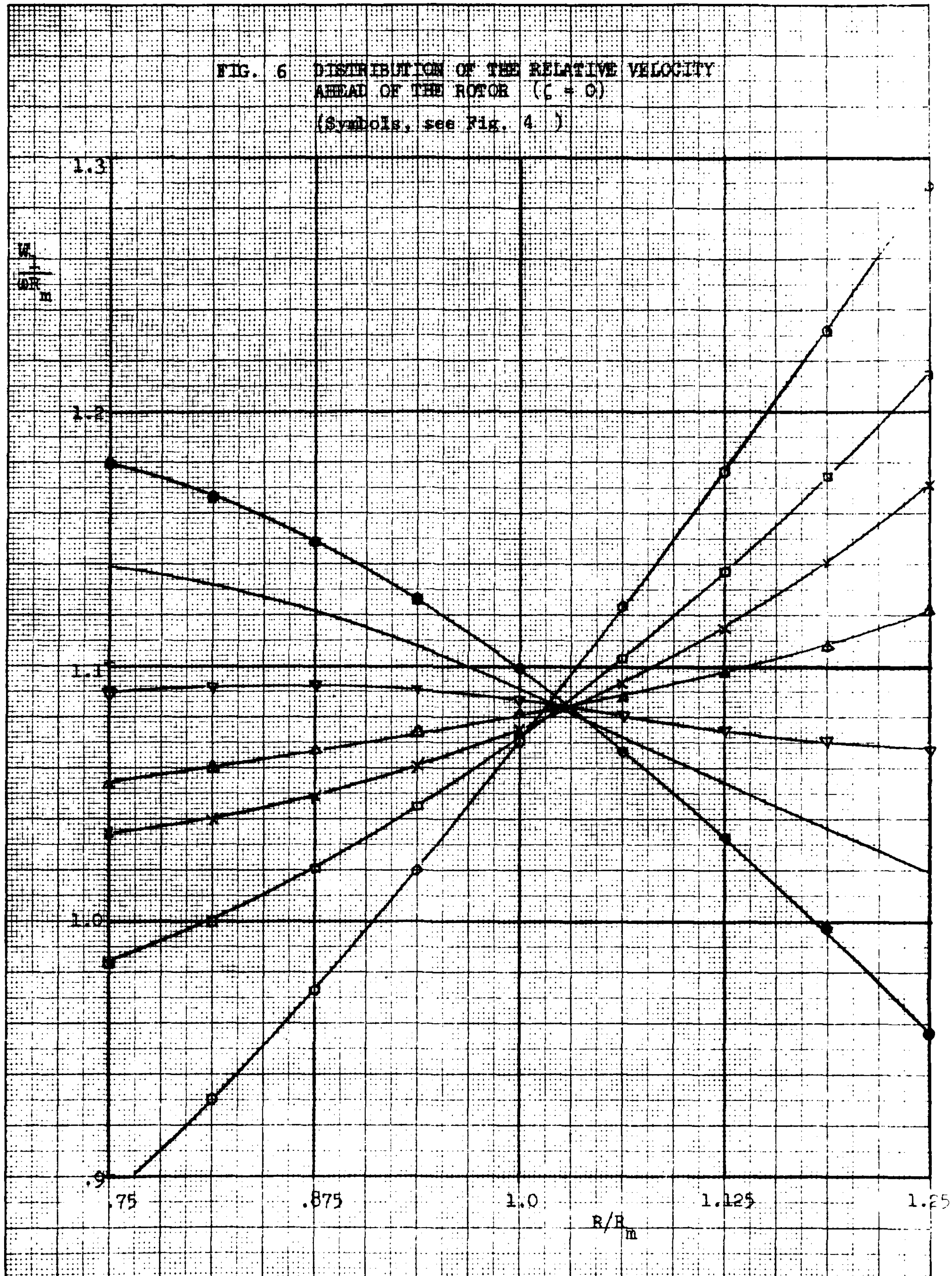


FIG. 6 DISTRIBUTION OF THE RELATIVE VELOCITY
AHEAD OF THE ROTOR (ζ = 0)
(Symbols, see Fig. 4)
$\frac{W_1}{\omega R_m}$
1.3
1.2
1.1
1.0
.9
.75
.875
1.0
1.125
1.25
R/R$_m$

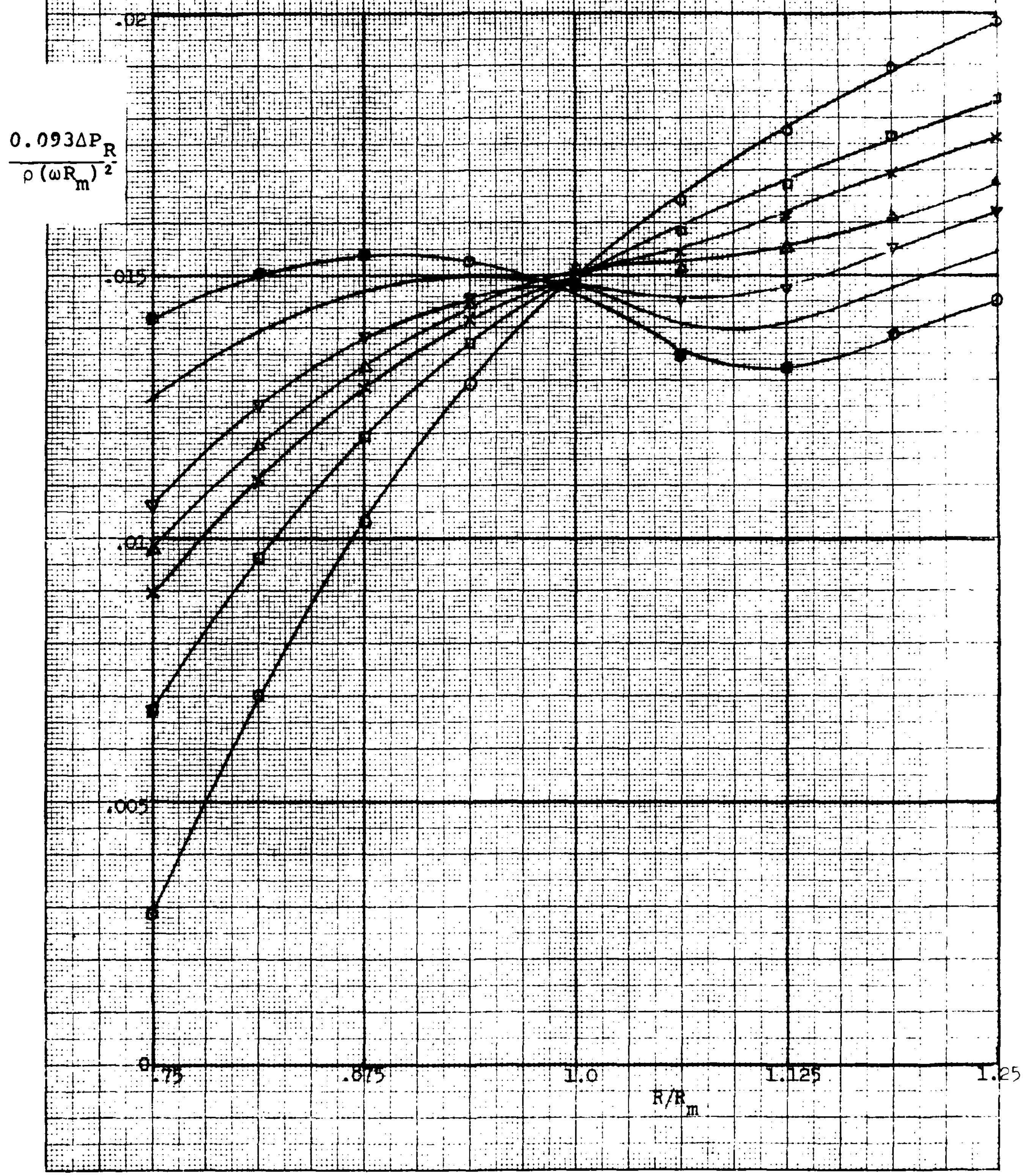
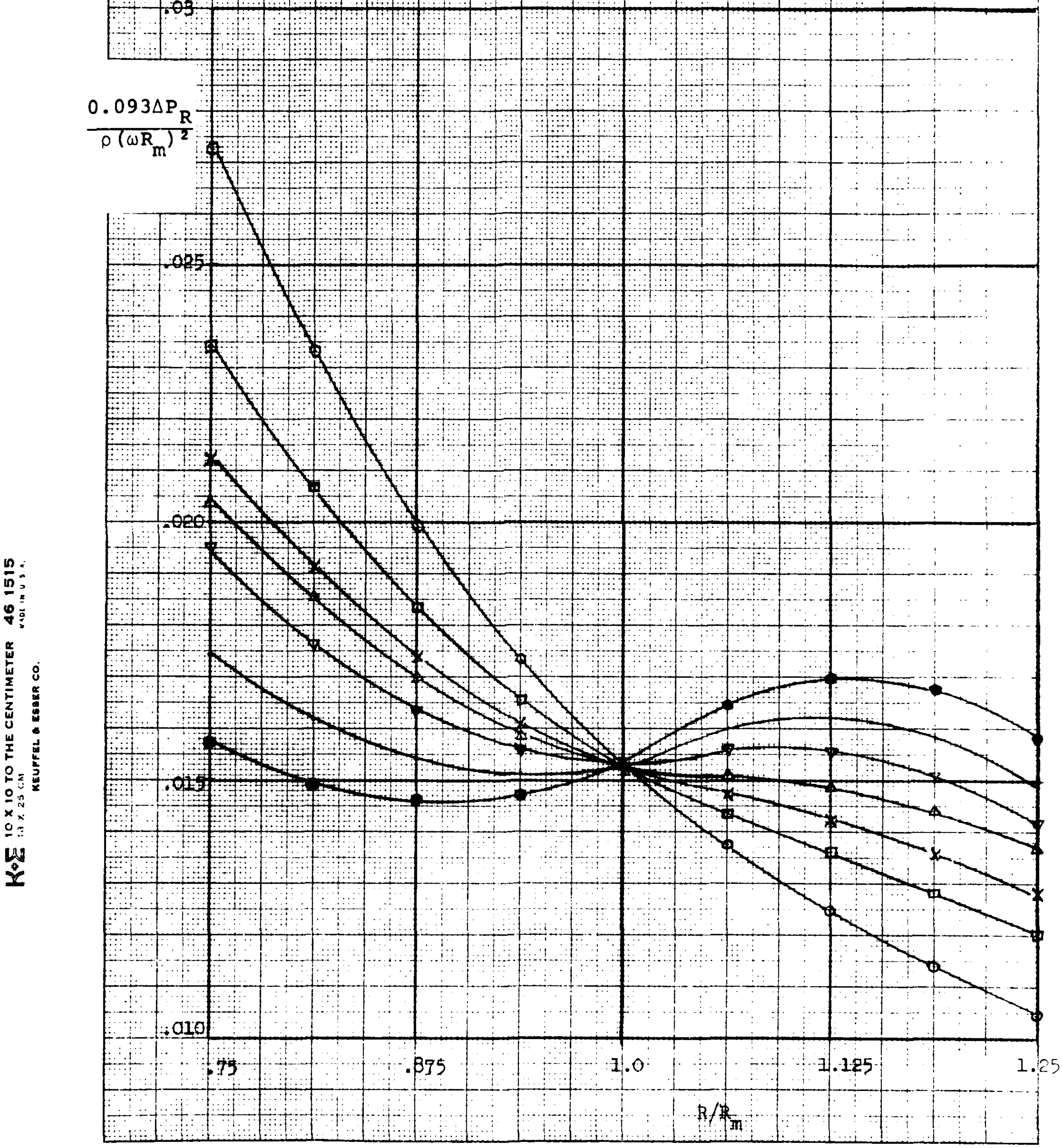


FIG. 7a INCREASE OF STATIC PRESSURE IN THE ROTOR ($\zeta = 0$)
(Symbols, see Fig. 4)

FIG. 7b INCREASE OF STATIC PRESSURE IN THE STATOR
($\zeta = 0$)
(Symbols, see Fig. 4)



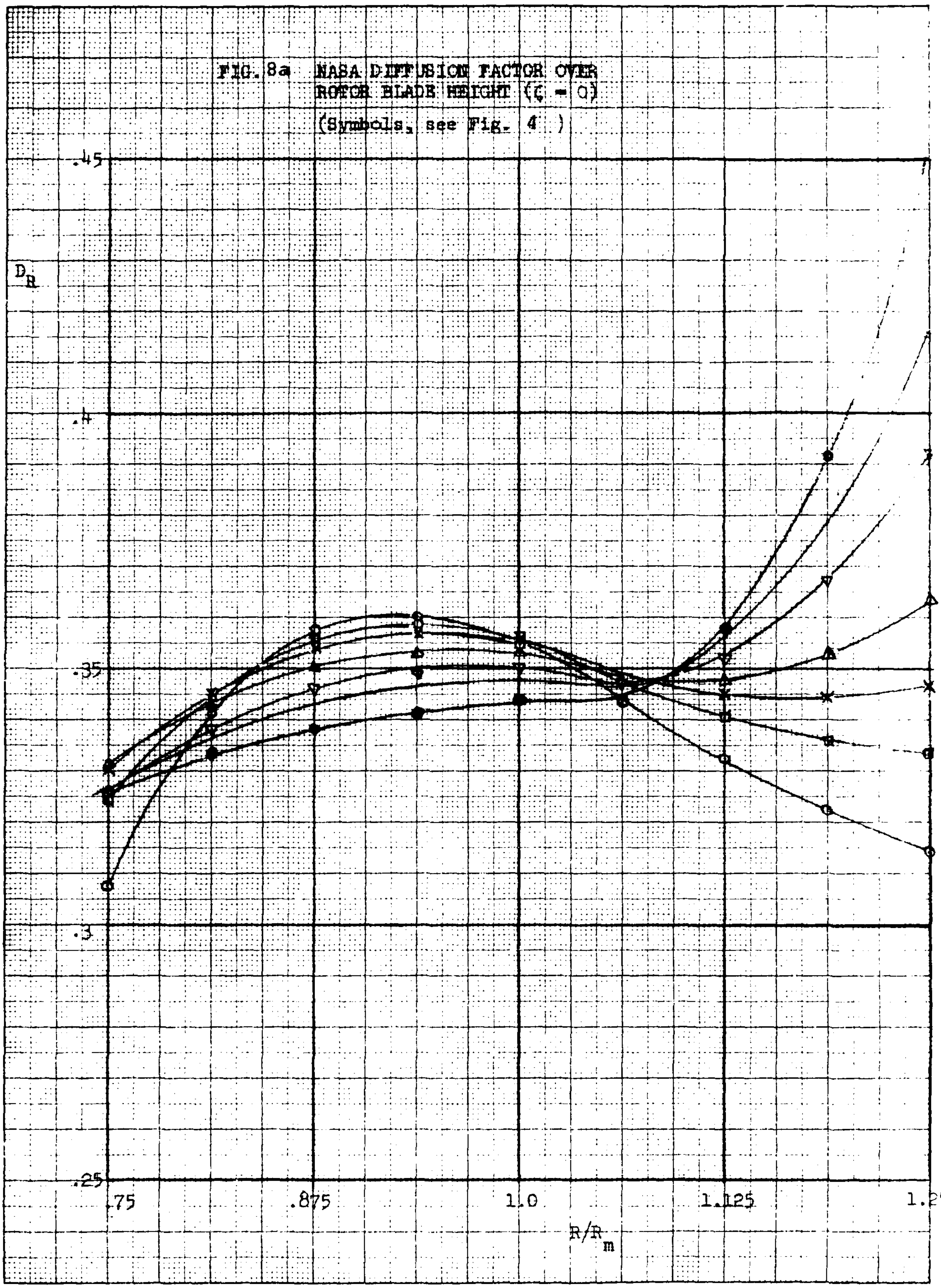


FIG. 8a NASA DIFFUSION FACTOR OVER ROTOR BLADE HEIGHT (ζ = 0)
(Symbols, see Fig. 4)
D_R
.45
.4
.35
.3
.25
.75
.875
1.0
1.125
1.25
R/R_m

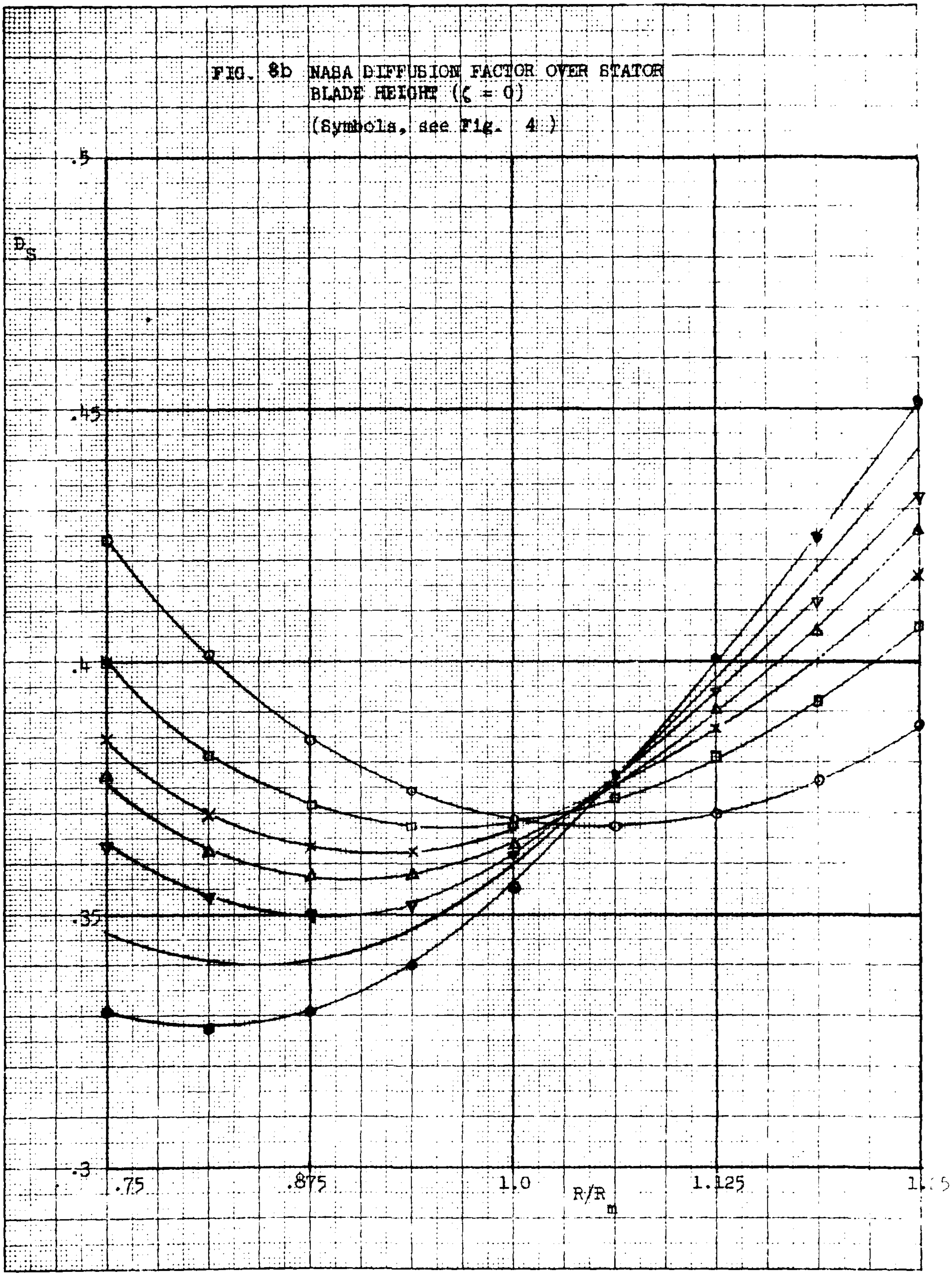


FIG. 8b NASA DIFFUSION FACTOR OVER STATOR BLADE HEIGHT (ζ = 0)
(Symbols, see Fig. 4)
D_S
.5
.45
.4
.35
.3
.75
.875
1.0
1.125
R/R_m

FIG. 9a AXIAL VELOCITY PROFILE AHEAD OF THE ROTOR, WITH IMPOSED ENTHALPY GRADIENT $\zeta = +0.15$

(Symbols, see Fig. 4)

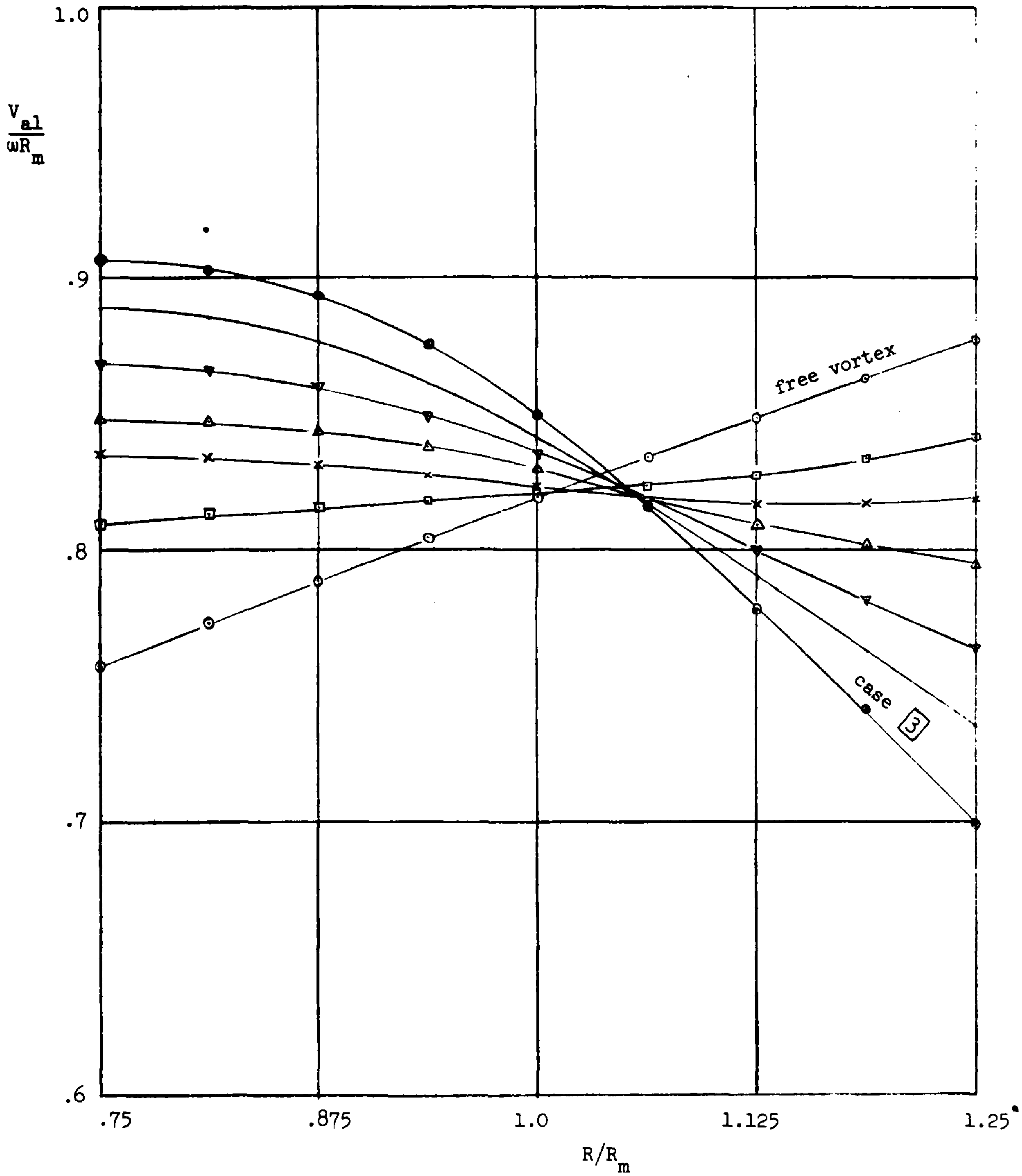


FIG. 9b AXIAL VELOCITY PROFILE AFTER THE ROTOR, WITH IMPOSED ENTHALPY GRADIENT ζ = +0.15

(Symbols, see Fig. 4)

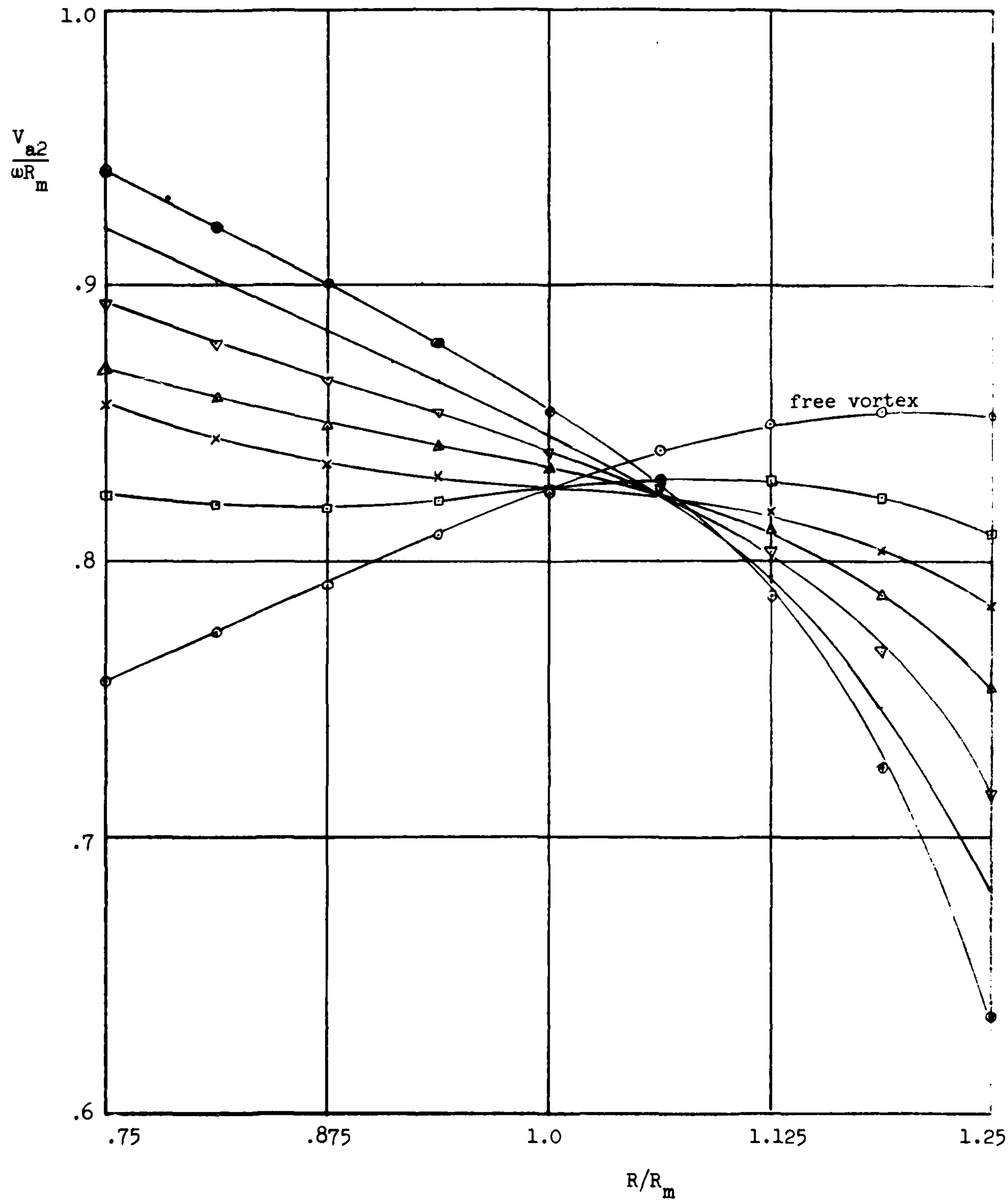


FIG. 10 RELATIVE VELOCITY AHEAD OF THE ROTOR, WITH IMPOSED ENTHALPY GRADIENT $\zeta = +0.15$
(Symbols, see Fig. 4)

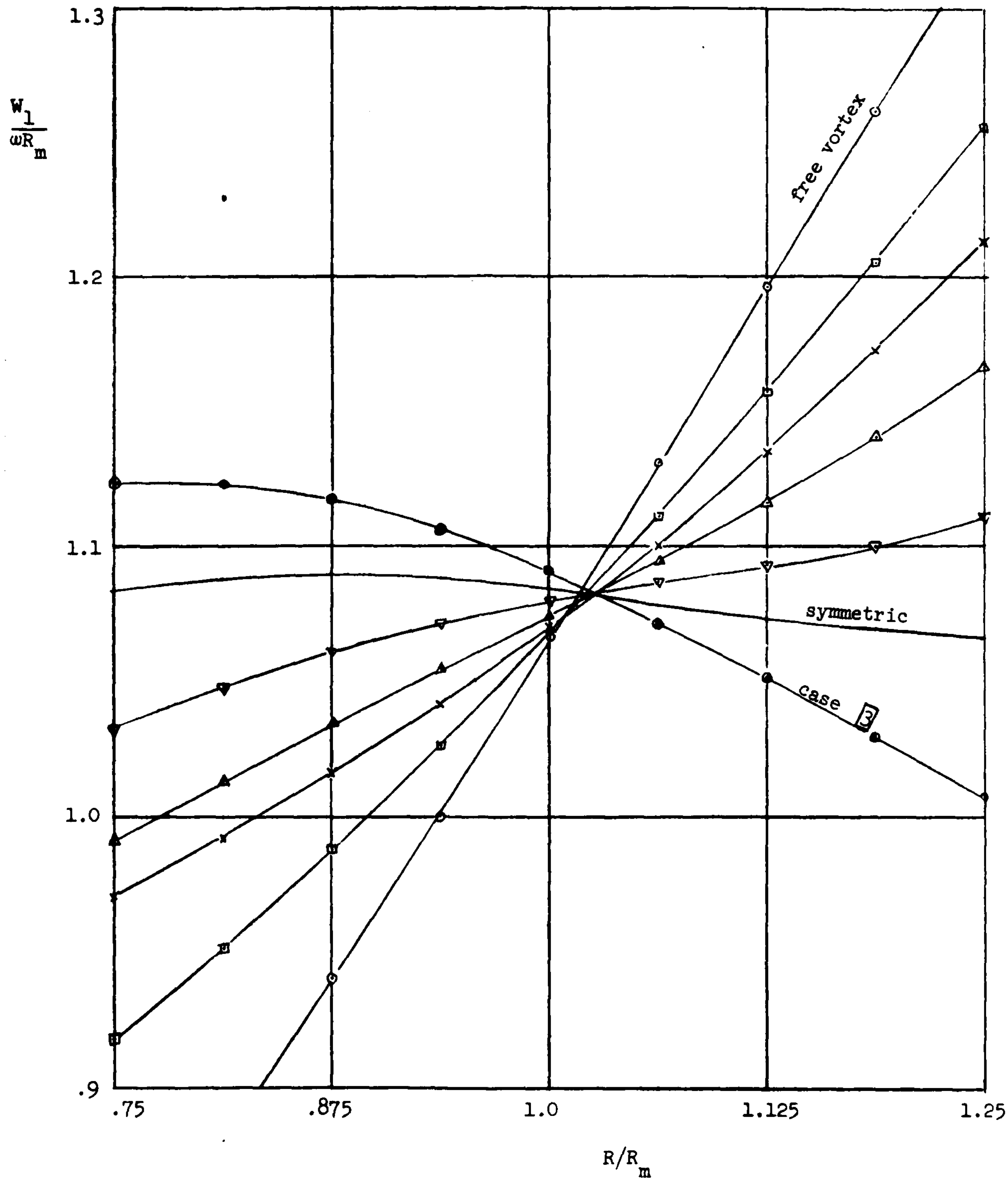


FIG. 11a NACA DIFFUSION FACTOR OVER ROTOR BLADE HEIGHT, WITH IMPOSED ENTHALPY GRADIENT ζ = +0.15

(Symbols, see Fig. 4)

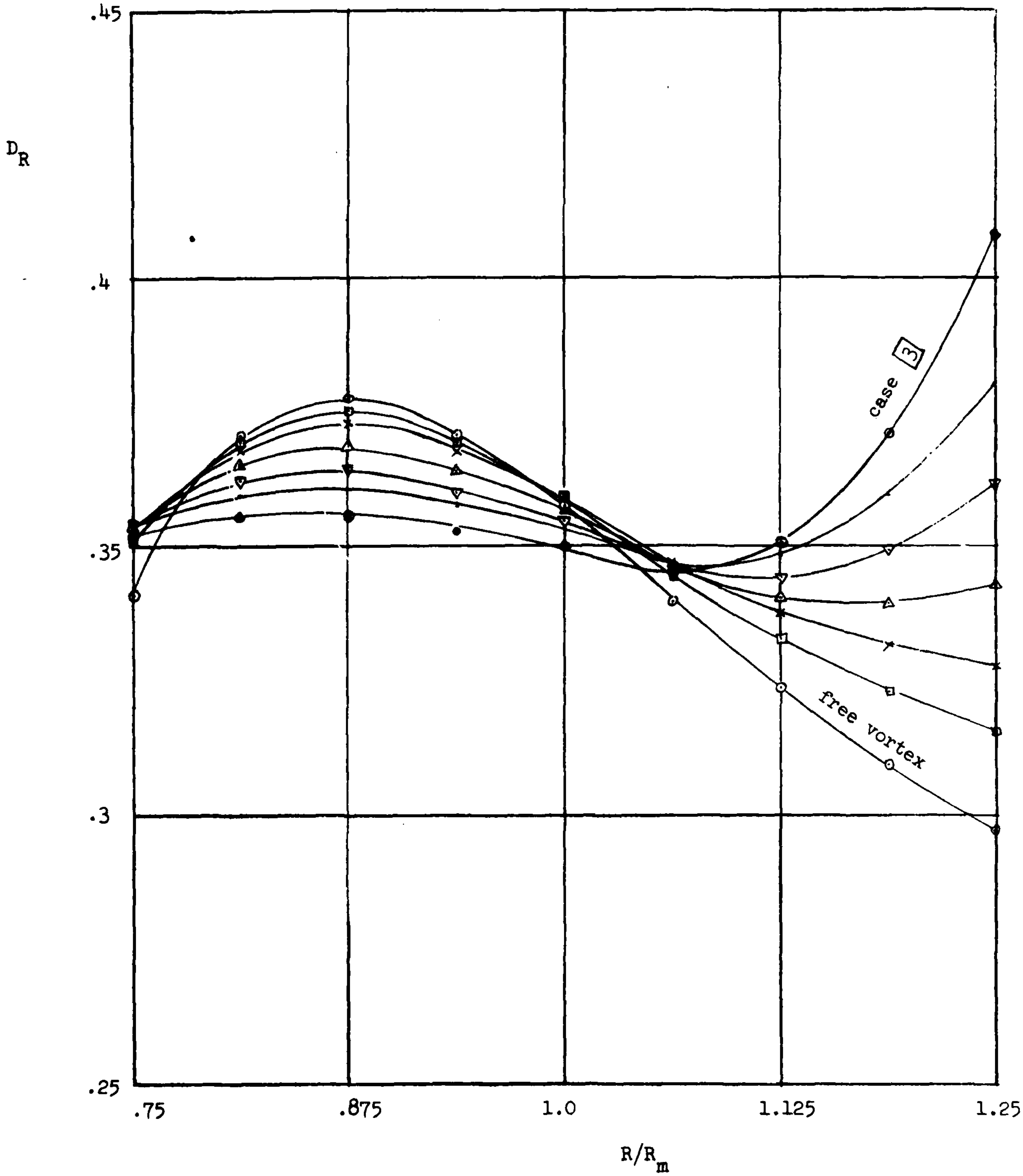


FIG.11b NACA DIFFUSION FACTOR OVER STATOR BLADE HEIGHT, WITH IMPOSED ENTHALPY GRADIENT ζ = +0.15

(Symbols, see Fig. 4)

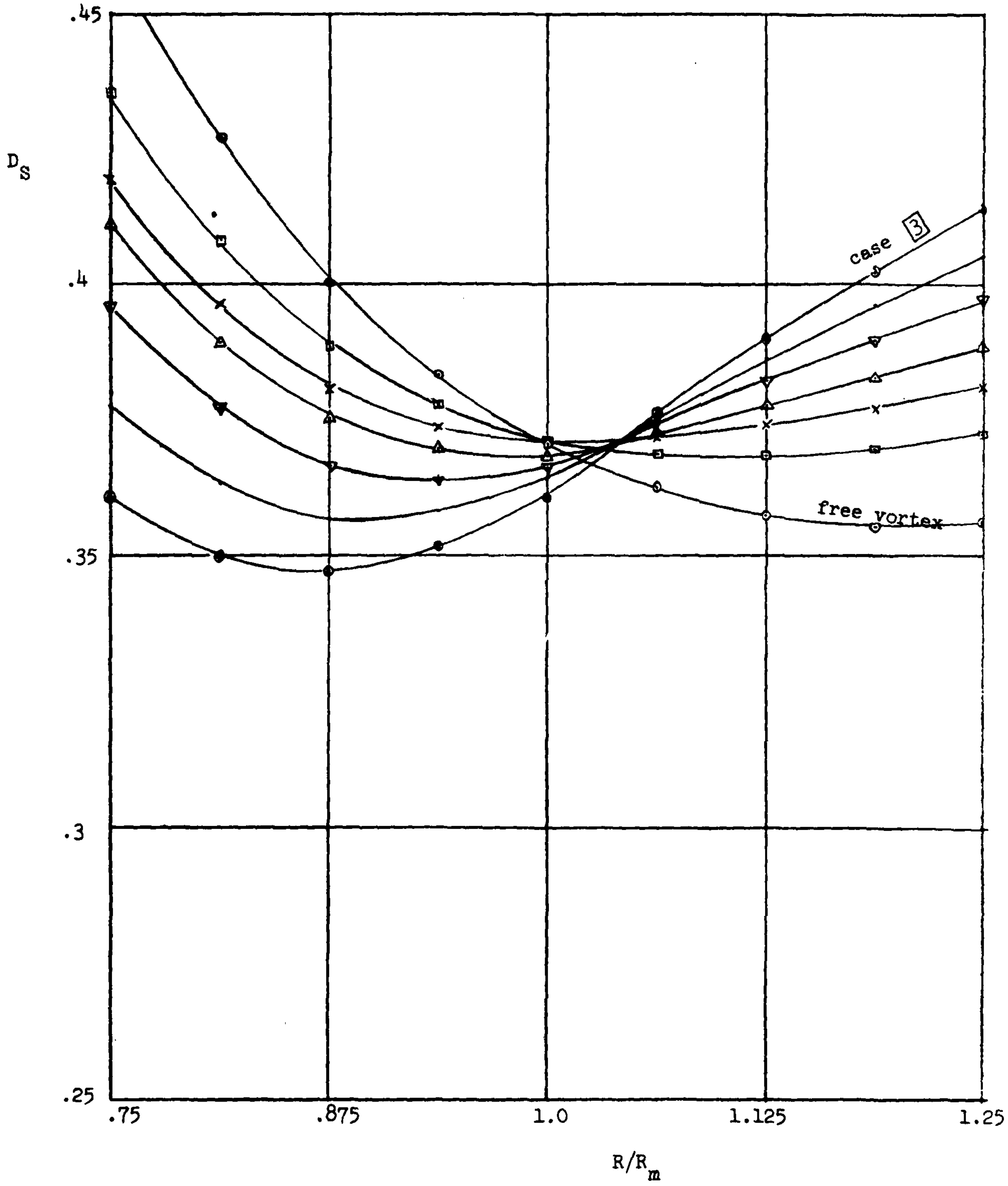


FIG. 12 INCREASE OF STATIC PRESSURE IN THE ROTOR, WITH IMPOSED ENTHALPY GRADIENT $\zeta = +0.15$

(Symbols, see Fig. 4)

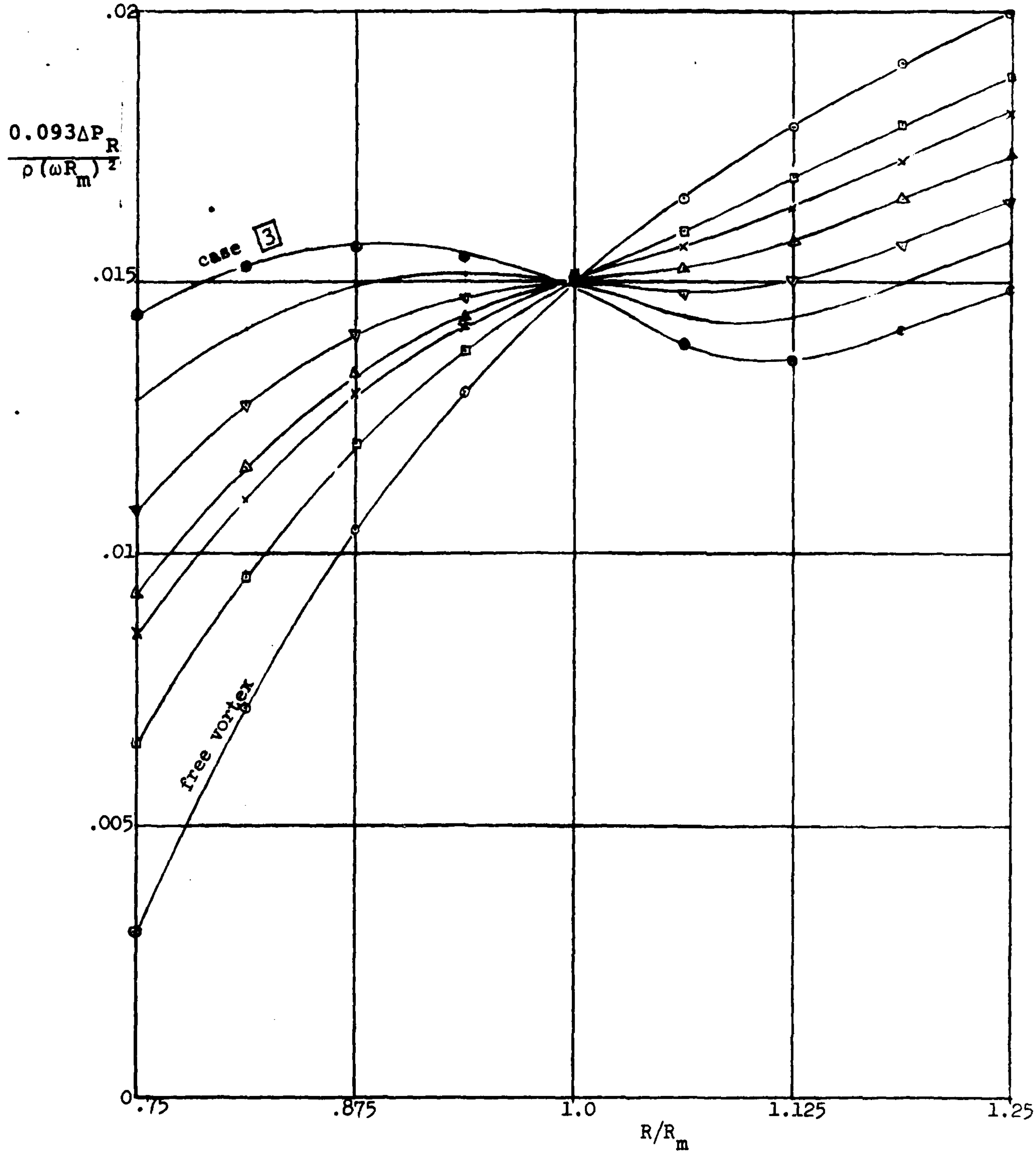


FIG. 13 INFLUENCE OF THE IMPOSED ENTHALPY GRADIENT ON THE RELATIVE VELOCITY AT THE ROTOR TIP

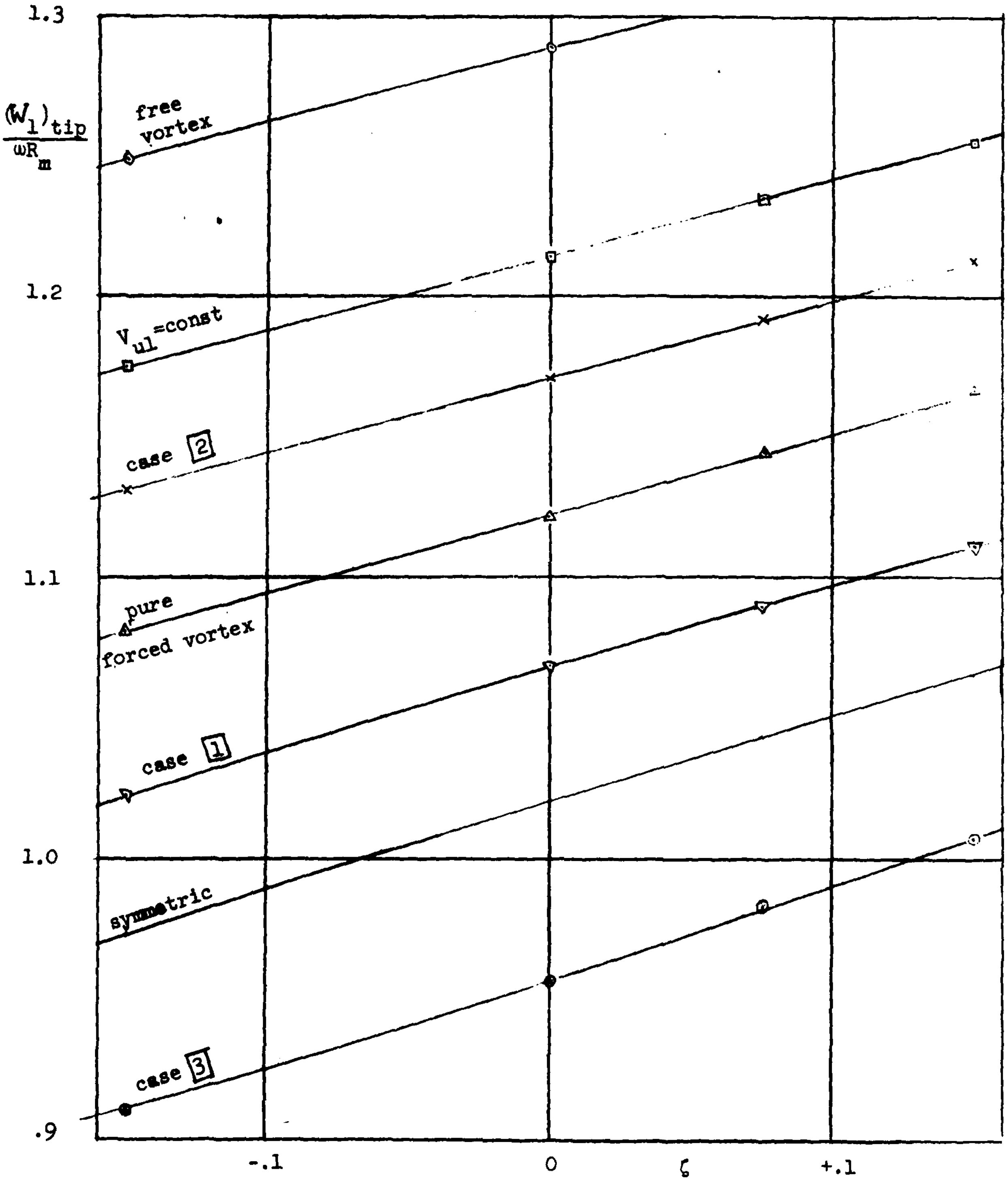


FIG. 14 INFLUENCE OF IMPOSED ENTHALPY GRADIENT ON THE NACA DIFFUSION FACTOR AT ROTOR TIP

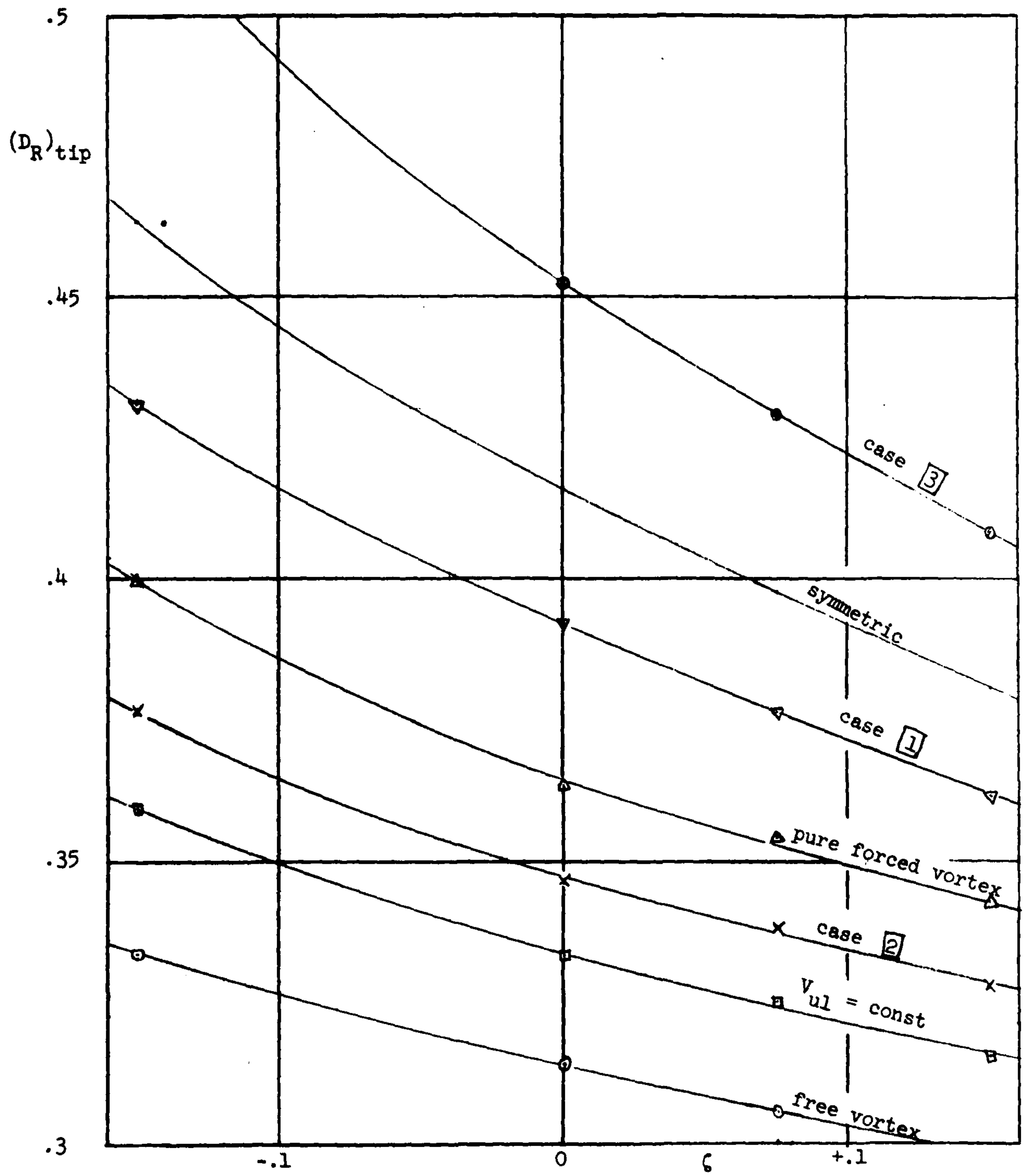


FIG. 15a: INFLUENCE OF THE IMPOSED ENTHALPY GRADIENT ON THE NACA DIFFUSION FACTOR AT STATOR HUB

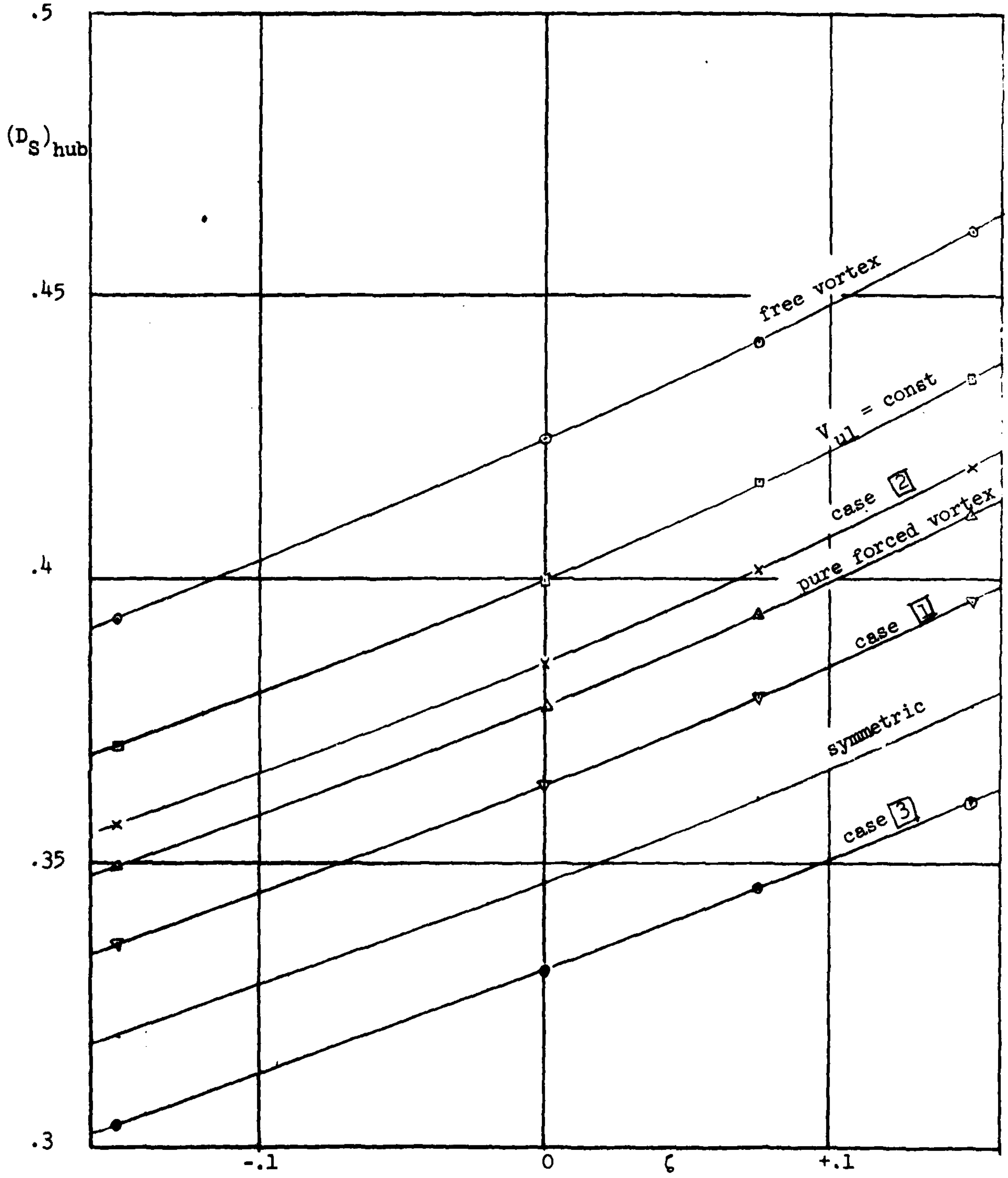


FIG. 15b INFLUENCE OF THE IMPOSED ENTHALPY GRADIENT ON THE NACA DIFFUSION FACTOR AT STATOR TIP

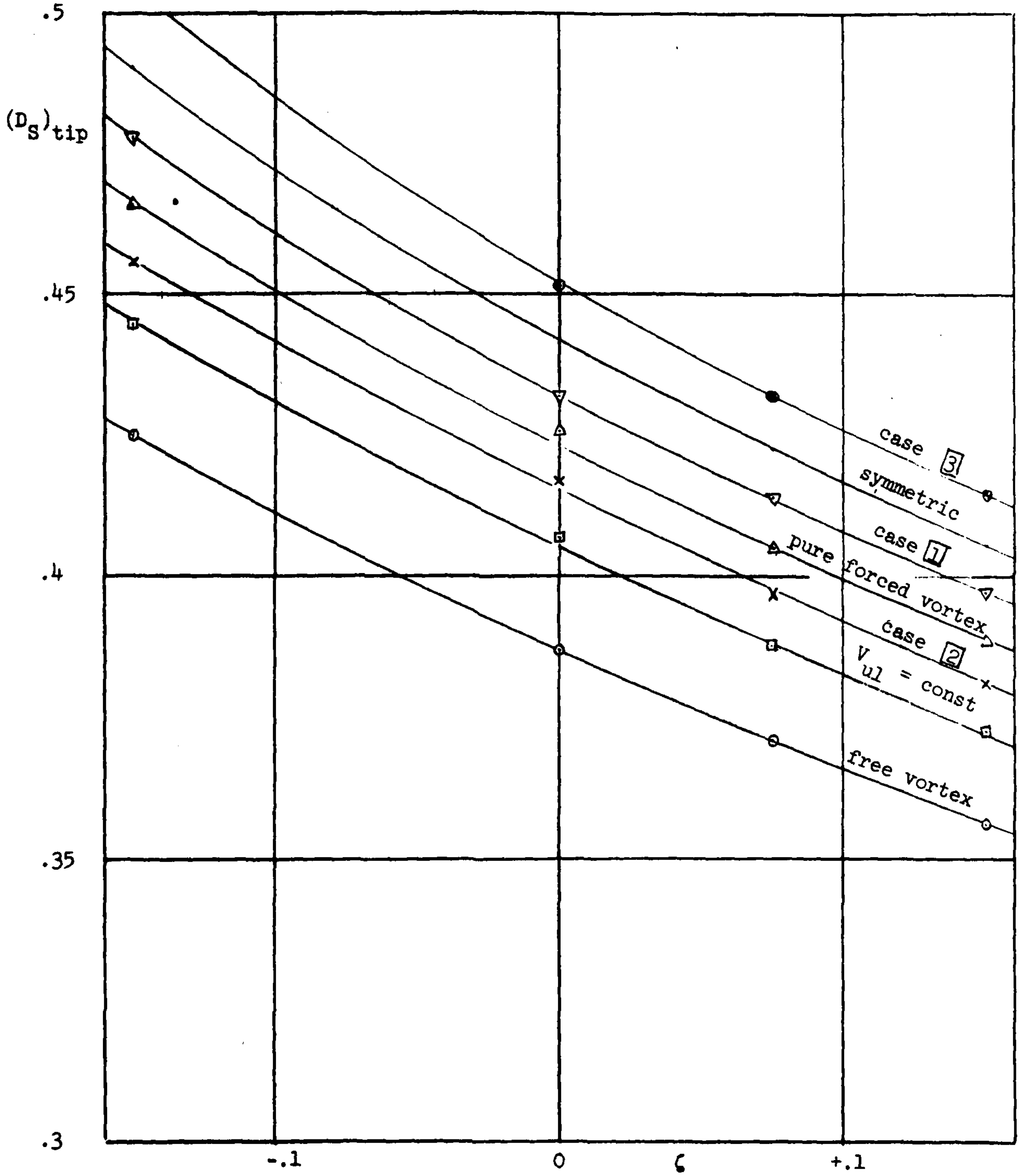
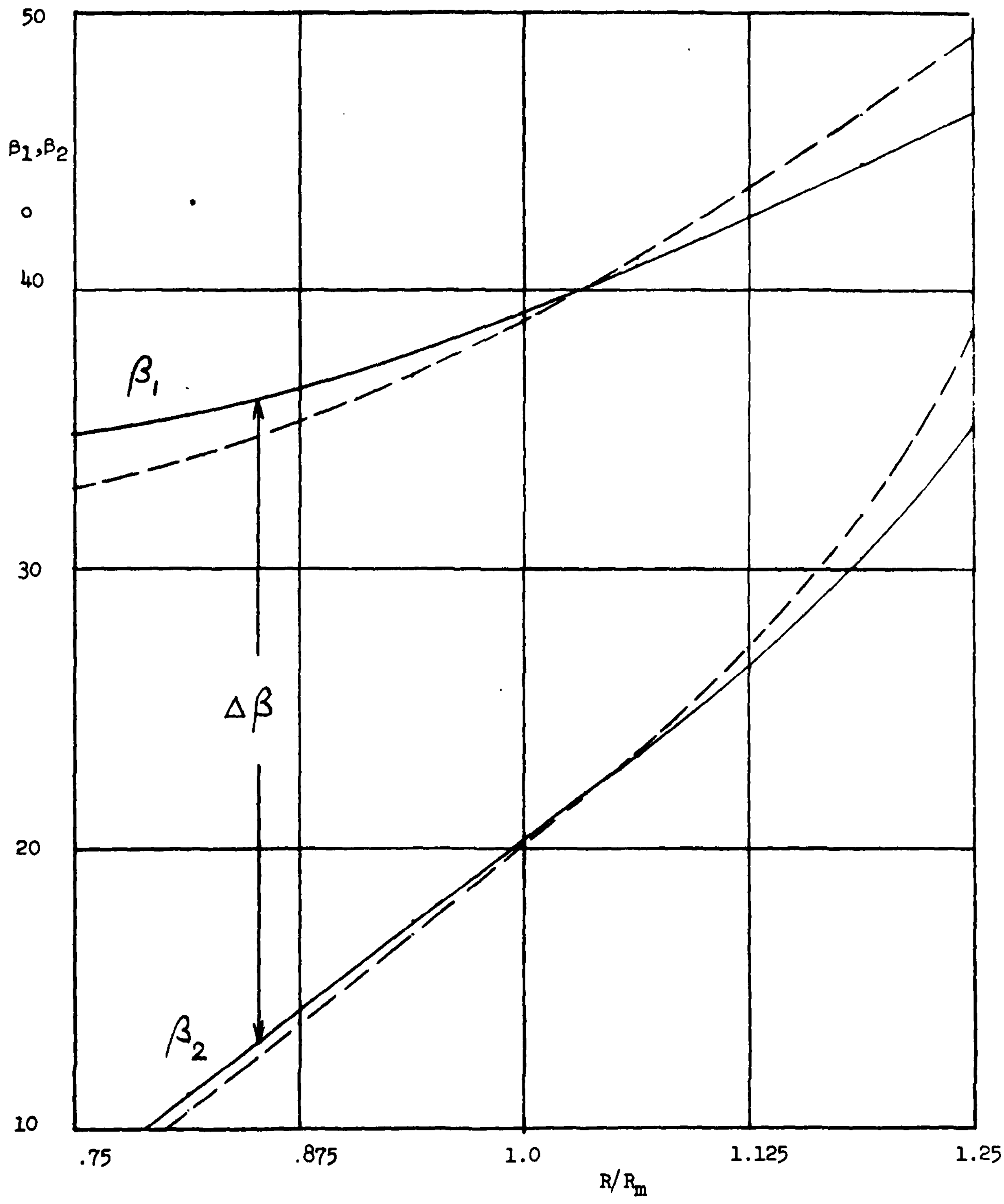


FIG. 16 RELATIVE FLOW ANGLES AHEAD (β_1) AND AFTER THE ROTOR (β_2) FOR SYMMETRICAL BLADING
— with positive enthalpy gradient (ζ = +0.15)
-- without enthalpy gradient (ζ = 0)



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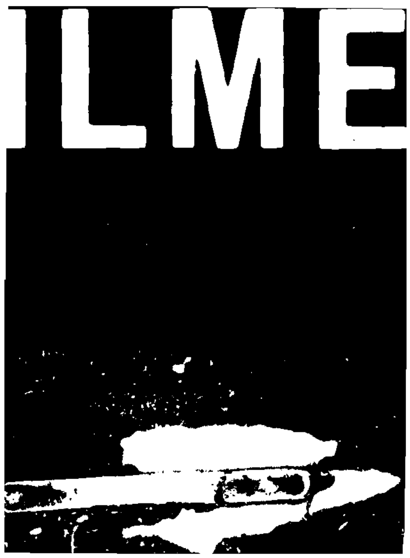
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